

The Effects of Compression Ratios and Fuels on the Energetic, Exergetic and Ecological Efficiency of an Air Standard Otto Cycle

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Abstract

In this study, the effect of the change in compression ratios, fuels and variable specific heat on the energetic, exergetic, and ecological efficiency of methanol, ethanol, iso-octane & liquefied petroleum gas (LPG) is being investigated in SI engine cycle model. The cycle model consists of compression, combustion, expansion, exhaust and internal heat transfer processes. First and second laws of thermodynamics principle are applied to the cycle model to perform energy and exergy analysis at stoichiometric condition. The exergy of the fuel is being distributed in exergy with internal heat transfer, exergy with net work transfer, irreversibilities during combustion, exhaust gases, compression and expansion processes. The result showed that the first law efficiency, second law efficiency, ecological efficiency, exergy with net work and exergy destruction during compression and expansion process increases with the increase in compression ratio for the all investigated fuels. The results also showed that the irreversibilities during combustion, internal heat transfer and exhaust gases decreases with the increase in compression ratio for all the examined fuels. The specific heats of all the fuels increases from compression to the combustion and decreases slightly during exhaust stroke.

Keywords

Energetic Efficiency; Exergetic Efficiency; Ecological Efficiency; Compression Ratio

Introduction

The increased importance of efficient utilization of fossil fuels in engines with reduced emissions continues to drive the need for thorough understanding of thermodynamics of internal combustion engines. Traditional analysis of thermodynamics efficiency of internal combustion engines is performed by applying a mass and energy balance throughout the engine cycle

according to the first law of thermodynamics [1, 2, 3] and comprehensive analysis of idealized internal combustion engines are found in the literature, these studies have served the basis for further optimization of current engine technologies [4]. However, practical systems are far from the thermal efficiencies predicted by the first law because analysis based on first law is often fails to identify the deviation from ideality [5, 6, 7].

The deviation between the efficiency predicted by the first law and the observed efficiency of the engine is an evidence of the defective exploitation of the fuel, which is due to the presence of internal thermodynamic irreversibilities in the combustion process. First law of thermodynamics can not identify and quantify the sources of losses (irreversibilities) in the engine, it simply provide the overall efficiency and hence fails to clarify the reasons of deviation between the ideal and actual performance of the engine [8,9]. On the other hand, the second law of thermodynamics offers a new perspective for the analysis of the performance of the engine based on the concept of exergy. It clearly indicates the presence of irreversibilities with combustion and heat transfer process and allows a thermodynamic evaluation of energy conservation option in internal combustion engine cycles. Reduction of irreversibilities in internal combustion engine cycles leads to the effective exploitation of fuels. One of the major advantages of exergy analysis is that the result distinguishes energy with the potential to produce work from energy at or near the dead state with little potential to produce work. By understanding the differences, more informed decisions may be obtained concerning the various trade off engine operation and design. Further new or unique

engine concepts may be evaluated from a more comprehensive perspective [10, 11, 12].

Many works are available in the literature advocating the importance of exergy based analysis for the performance evaluation of internal combustion engine [13, 14, 15, 16]. These studies continuously shows that four terms dominant in exergy balance as applied to the engine cylinder (excluding sub system or compounding) (a) Exergy destruction due to combustion; (b) Exergy transfer via heat loss due to the finite temperature difference between the combustion zone and walls of the cylinder; (c) exergy lost due to exhaust of combustion products; and work transfer due to gas expansion. The exergy destruction due to mixing of residual gases with intake fuel air mixture and pumping losses are much smaller than the above magnitudes. These studies clearly shows that a careful management of exergy destruction and exergy transfer is required for the effective exploitation of the fuel and hence to achieve the goals of clean energy efficiency benefits and reduced emissions.

Always there are some differences between real air – fuel cycles and ideal air standard cycles, therefore the results from air standard thermodynamic analysis will deviate from actual conditions and property values of temperature and pressure are very representative of actual engine values, depending on the geometry and operating conditions of the real engine[17]. Therefore, to make the analysis of the engine cycle much more manageable, air standard cycles are used to describe the major processes occurring in internal combustion engines. In practice, air standard analysis is important for illustrating the thermodynamic aspects of an engine. Meanwhile, it can provide approximate estimates of trends as major engine operating engine variables change. Characteristic of power and efficiency for Otto, dual, and Miller cycles were derived with consideration of heat transfer and friction like term losses [18, 19]. The effects of heat transfer through a cylinder wall on the work output were examined for Otto and Diesel cycles [20]. However, Ozsoysal's study was only focused on the temperature limitations and no performance analysis was presented without considering the effects of variable specific heats of the working fluids[21]. In particular, no performance analysis is available in the literature with emphasis on the Otto cycle with consideration of variable specific heats of the working

fluids, second law analysis, ecological efficiency and internal irreversibility's as a percentage of fuel chemical exergy. This study is aimed at analyzing these effects (i.e. variable specific heat of working fluids, first and second law analysis and ecological efficiency) with the change in compression ratio for the ethanol, methanol, iso-octane and LPG fuels.

SI Engine Cycle Description

Fig.1 shows the temperature entropy diagram of the air standard Otto cycle with the internal irreversibility's. Thermodynamic cycle 1-2s-3-4s-1 denotes the air standard Otto cycle without internal irreversibility's while cycle 1-2-3-4-1 designates the air standard Otto cycle with internal irreversibility's. The cycle considered for analysis is a complete representation of the four stroke SI engine including the compression, combustion, expansion and exhaust processes and as shown in Figure.1. Process 1-2s is a reversible adiabatic compression, while process 1-2 is an irreversible adiabatic process that takes into account the internal irreversibility's in the real compression process. The heat addition is a constant volume process 2-3. Process 3-4s is a reversible adiabatic expansion, while 3-4 is an irreversible adiabatic process that takes into account the internal irreversibility's in the real expansion process. The heat rejection is at a constant volume process 4-1. We relax the assumptions that there are no irreversibility's during various processes of the cycle and specific heats of the working fluids are constant. In other words, irreversibility's during compression, combustion, expansion and exhaust processes are characterized by a percentage of fuel's chemical exergy. Furthermore, we consider the variable specific heats of the working fluids that is significant in practice cycle analysis.

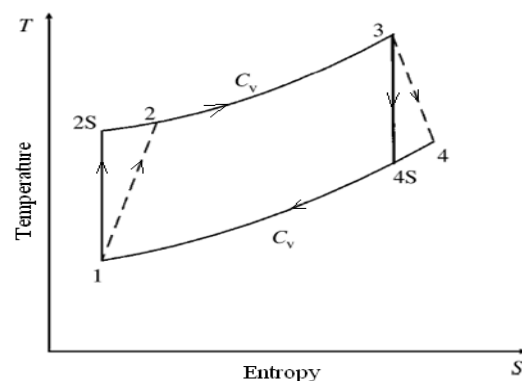


FIG. 1 (FOUR STROKE SI CYCLE)

TABLE 1 COMPARISON OF SELECTED FUELS PROPERTIES

Property	Methanol	Ethanol	Iso-octane	Propane	Butane
Chemical formula	CH ₃ OH	C ₂ H ₅ OH	C ₈ H ₁₈	C ₃ H ₈	C ₄ H ₁₀
Molecular weight(Kg/kmol)	32.04	46.07	114.228	44.14	58.17
Oxygen present (wt %)	49.9	34.8	-	-	-
Density (g cm ⁻³)	792	789	700	2.458×10 ⁻³	1.865×10 ⁻³
Freezing point at 1 atm (°C)	-97.778	-80.0	-107.378	-0.5	-42
Boiling temperature at 1 atm (°C)	64.9	74.4	99.224	231	273
Auto-ignition temperature(°C)	463.889	422.778	257.23	482	405
Latent heat of vaporization at 20°C (KJ/Kg)	1103	840	349	46.3	45.72
Stoichiometric air/fuel ratio (AFR)	6.47	9.0	15.2	15.6	15.34
Lower heating value of the fuel (KJ/Kg)	20000	26900	44300	46350	45710
Research octane number (RON)	111	108	100	105	92
Motor octane number (MON)	92	92	100	104	89

Assumptions:

1. The thermodynamic system is the closed chamber content.
2. The cylinder content is assumed to be spatially homogeneous and to occupy one zone.
3. The fuel is assumed to be completely vaporized and mixed with the reactant air.
4. The thermodynamic properties (including pressure and temperature) are spatially uniform.
5. Variation of specific heat with temperature at different process points has been incorporated.
6. Combustion takes place instantaneously and it is assumed that there is no heat loss from combustion chamber to the cylinder wall during combustion.
7. Compression and Expansion processes are polytropic.

Thermodynamic Engine Model

An entropy generation analysis was applied to SI engine and an entropy generation calculation model was

developed. Mathematical formulation of the principle of the non-conservation of entropy for a non- steady flow process in a time interval t_1 to t_2 following the second law of thermodynamics can be written as

$$\sum_i m_i s_i - \sum_e m_e s_e + \sum_r \left(\frac{Q_r}{T_r} \right)_{1-2} + S_{\text{gen}, 1-2} = S_2 - S_1 \quad (1)$$

Where m_i and m_e respectively denote the amount of mass across input port i and exiting across port e ; $(Q_r)_{1-2}$ denotes the amount of heat transferred into the control volume r on the control surface. $S_{\text{gen}, 1-2}$ denotes the amount of entropy generated in the control volume and S_1 and S_2 are the amounts of entropy in the control volume at t_1 and t_2 .

Fuel properties of methanol, ethanol, Isooctane and LPG (A mixture of 75 % propane and 25 % butane) are given in table 1.0 [22, 23, 24, 25].

The entropy generation during the four key processes of the SI engine is calculated after applying the equation (1) and the consequent exergy destruction is evaluated after using the Gouy –Stodola theorem as

$$E_{\text{dest}} = T_0 S_{\text{gen}} \quad (2)$$

The molecular weight of the products and the reactants were found by using the following relations

$$M_R = \sum n_i M_i / N \quad (3)$$

Where n_i is the number of moles of the species in the reactant or product per unit mass of the working fluid. M_i is the molecular mass of the species and N is the total number of moles.

The specific heat of the product and the reactant are found by using the relation

$$C_p = \sum n_i C_{pi} \quad (4)$$

$$C_v = \sum n_i C_{vi} \quad (5)$$

$$R = C_p - C_v \quad (6)$$

The value of C_p and C_v were calculated at three points:

- (1) At the beginning of the compression process.
- (2) During combustion process at the average temperature (temperature after compression + adiabatic flame temperature) / 2.
- (3) At the end of the expansion process.

The values of C_p and C_v of different fuels are calculated using the following equations. Specific heat is a function of temperature and it is represented as

$$C_p = a + bT + cT^2 + dT^3 \quad (T \text{ in K, } C_p \text{ in kJ / kmol. k}) \quad (7)$$

Substance	a	b	c	d
Methanol	19.0	9.152×10^{-2}	-1.22×10^{-5}	-8.039×10^{-9}
Ethanol	19.9	20.96×10^{-2}	-10.38×10^{-5}	20.05×10^{-9}
Iso-octane	-0.053	6.75×10^{-3}	-3.67×10^{-6}	-0.39×10^{-9}
Propane	-4.04	30.48×10^{-2}	-15.72×10^{-5}	31.74×10^{-9}

Process (1-2)

The fuel air mixture (charge) is assumed to be an ideal gas; the entropy generation during the compression process is given by:

$$S_{gen} = m (C_p \ln T_2/T_1 - R \ln P_2/P_1 + Q / T_o) \quad (8)$$

Where T_2 is the charge temperature at the end of the compression and it is given by

$$T_2/T_1 = (V_1/V_2)^{n-1} = (r)^{n-1} \quad (9)$$

Where V_1 is the cylinder volume, V_2 is the clearance volume, r is the compression ratio and n is the polytropic index. The value of n is taken as 1.348, 1.365, 1.354 and 1.363 for ethanol, methanol, iso-octane and LPG during compression process and 1.24 during expansion process for all the selected fuels.

Process (2-3)

The exergy destruction during the process of heat addition of constant volume (from heat source to the fuel) is given by

$$e_{Dest,2-3} = T_o (C_v \ln T_3/T_2 - Q_{in}/T_{source}) \quad (10)$$

Where Q_{in} is the amount of heat liberated by the combustion of the fuel and it is being simulated as supplied by the outside source at a temperature of T_3 which is different for different fuels and different at different compression ratios.

Where $Q_{in} = m_f \times \text{LHV}$ of the fuel and

$$m_f \times \text{LHV} \times \eta_{comb} = m C_v (T_3 - T_2) \quad (11)$$

Where, m is the mass of the air fuel mixture and combustion efficiency is assumed 95% at stoichiometric condition.

The exergy destruction during the subsequent combustion, which is the most concerned irreversible process, may be obtained after applying the exergy balance over the engine combustion chamber as

$$e_2 + e_Q = e_3 + e_{Dest,comb} \quad (12)$$

Where e_Q is the amount of exergy associated with the heat transfer q_{in} and is given by

$$e_Q = (1 - T_o/T) m_f \times \text{LHV} \quad (13)$$

e_2 is the physical exergy of the charge before combustion, and e_3 is the sum of physical and chemical exergy of the fuel air mixture after combustion. The equation (8) may be further elaborated as

$$e_{Dest,comb} = e_{2,physical} - e_{3,physical} - e_{3,chemical} + e_Q \quad (14)$$

$$e_{Dest,comb} = (e_2 - e_3)_{ph} - e_{3ch} + e_Q$$

the term $(e_2 - e_3)_{ph}$ physical for a closed system engine combustion chamber is given by

$$(e_2 - e_3)_{ph} = u_2 - u_3 + p_o(v_2 - v_3) - T_o(s_2 - s_3) \quad (15)$$

and the term $e_{3,chemical}$ which is the chemical exergy of the fuel is given by sum of the first term of the equation (11) is the reactive chemical exergy which is the work produced due to reversible transition of the species of the mixture from restricted dead state to the ultimate dead state (environmental state), and the third term of the equation (11) is the diffuse chemical exergy which is the maximum work produced due to the difference of partial pressures of the ambient species of mixture

between the restricted dead state and the partial pressures of the same species at the environment state.

$$E_{3, \text{chemical}} = E_{\text{Re}} + E_{\text{Diff}}$$

$$E_{3, \text{chemical}} = \sum_{i=1}^n N_i g_i(T_0 p_i) - \sum_{j=1}^n N_j g_j(T_0 p_j) + \sum_{j=1}^n N_j \bar{R} T_0 \ln p_j / p_{j0} \quad (16)$$

The specific chemical exergy of the mixture $e_{3, \text{chemical}}$ may be obtained as:

$$E_{3, \text{chemical}} = E_{3, \text{chemical}} / \sum M_j N_j \quad (17)$$

Where M_j is the molecular weight and N_j is the number of moles of species j of the combustion products of the mixture.

In equation (16), the index i denotes the individual species present in the mixture at restricted dead state and the index j denotes the individual ambient species present in the mixture that were formed from the species i through a series of oxidation /reduction reactions, g_i and g_j are the Gibbs free energies, N_i and N_j are the number of moles, p_i and p_j are the partial pressure of the species denoted by the species i and j respectively. p_{j0} is the partial pressure of the ambient species of the environmental state and \bar{R} is the universal gas constant.

Process (3-4):

The exergy destruction during the expansion process (3-4) is calculated similar to the process of compression, and may be presented as

$$e_{\text{Dest, expansion}} = T_0 (C_p \ln T_4/T_3 - R \ln p_4/p_3) \quad (18)$$

Process (4-1)

The exergy lost during the exhaust process is calculated after using the entropy generation principle and is given by

$$\text{Exergy lost during exhaust} = T_0 (C_v \ln T_1/T_4 - Q_{\text{out}}/T_{\text{sink}}) \quad (19)$$

The term inside the bracket represents the entropy generation during the exhaust process and the value of C_v is taken corresponding to temperature T_4 . T_{sink} is assumed as 300 K. Q_{out} is the amount of heat carried with the exhaust gases which is given by

$$Q_{\text{out}} = m C_v (T_4 - T_1) \quad (20)$$

Where m is the mass of the exhaust gases

The First law efficiency or thermal efficiency of the SI engine is given by

$$\eta_{\text{th}} = \text{Net work done} / (m_f \times \text{LHV of the fuel}) \quad (21)$$

and the second law efficiency or exergy efficiency of the engine is given by

$$\eta_{\text{exergy}} = 1 - (\text{total exergy destruction during the cycle}) / \text{Chemical exergy of the fuel} \quad (22)$$

Environmental Consideration

The environmental impact from the use of fossil fuels and renewable fuels in internal combustion engines may be evaluated in terms of ecological efficiency, a concept which depends upon the environmental impact caused by CO_2 , SO_2 , NO_x and Particulate Matter (PM) emissions.

The effect of other pollutants may be related to the coefficients for the equivalent carbon dioxide $(\text{CO}_2)_e$ a hypothetical pollutant concentrations factor. For the calculations of this coefficients, the maximum observed value for the CO_2 concentration is divided by the corresponding air quality standard for CO_2 , SO_2 , NO_x and PM in 1 hour. The expression for $(\text{CO}_2)_e$ [26].

$$(\text{CO}_2)_e = \text{CO}_2 + 80(\text{SO}_2) + 50(\text{NO}_x) + 67(\text{PM}) \quad (23)$$

In eq (17) $(\text{SO}_2)_e = 80(\text{SO}_2)$ is the sulphur dioxide equivalent in $(\text{CO}_2)_e$, $(\text{NO}_x)_e = 50(\text{NO}_x)$ is the nitrogen dioxide equivalent in CO_2 , and particulate matter equivalent in CO_2 is $(\text{PM})_e = 67(\text{PM})$. The best fuel from the ecological point of view is that which presents minimum amount of $(\text{CO}_2)_e$ obtained from its burning. In order to quantify this environmental impact, the pollutant indicator (Π_g) is defined by

$$\Pi_g = (\text{CO}_2)_e / Q_i \quad (24)$$

Where $(\text{CO}_2)_e$ is taken in kg/kg of the fuel. Q_i is the fuel lower heating value (LHV), expressed in MJ/kg, and Π_g is expressed in kg/MJ.

Ecological Efficiency

The ecological efficiency is a parameter which indicates and allows the evaluation of the combustion engine performance in respect to the pollution emission by comparing the hypothetically integrated pollutant emissions (CO_2 equivalent emissions) to the existing air quality standards. The conversion efficiency is also

TABLE 2 COMPARISON OF THE POLLUTANT EMISSIONS FOR THE FUELS ANALYZED (AS PER BHARAT STAGE III)

Pollutant emissions	Methanol	Ethanol	Iso-octane	LPG
(CO ₂)e	1.7251	2.265	4.4329	3.36
CO ₂	1.37	1.91	3.02	2.99
NO _x	.003869	.00387	.004369	.00218
SO _x	.002	.002	.002	.00325
PM	2.57×10^{-5}	2.57×10^{-5}	.01544	2.57×10^{-5}

considered a determining factor on the specific emissions, expressed by a fraction number. The ecological efficiency may be determined after using (19) expression [26].

$$\epsilon = \left\{ \frac{204\eta_{th}}{\eta_{th} + \Pi_g} \ln(135 - \Pi_g) \right\}^{0.5} \quad (25)$$

The ecological efficiency ϵ is directly proportional to the thermal efficiency which may be taken as first law efficiency of the engine and inversely proportional to the Π_g , the pollutant indicator value and is also located between 0 and 1. The situation is considered unsatisfactory from the ecological point of view when $\epsilon = 0$ however $\epsilon = 1$ indicates the ideal situation from the point of view of energetic efficiency.

The pollutant indicator value of which is prerequisite to determine the ecological efficiencies values for internal combustion engine for the case of urban transport are calculated for the four selected fuels: Iso-octane, LPG, Ethanol and Methanol [27,28,29].

Results and Discussions

The effects of the (variable) type of fuel used and compression ratio of the engine are observed on the energetic, exergetic and ecological performance of SI engine. The effect of these two variables is also observed on the exergy destruction in the engine during: compression; combustion; heat transfer and exhaust process: The four fuels: Iso-octane, LPG, Ethanol and Methanol are selected and a suitable range of compression ratios for the operation of SI engine is chosen.

For the numerical appreciation of the results: The following parameters are used.

Variable specific heat, Exergy destruction during compression, Exergy lost via internal heat transfer, Exergy destruction during compression, Exergy destruction during expansion, Exergy destruction during exhaust process, Exergy with net work, First law efficiency, Second law efficiency, Brake specific fuel consumption, and Ecological efficiency.

Exergy Destruction during Compression

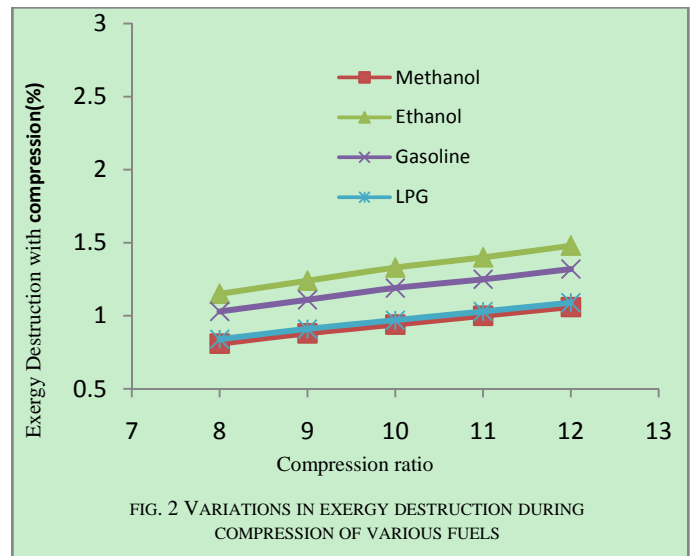


Fig. 2 shows the variation in exergy destruction during compression with the change in compression ratio for all selected fuels. It is shown that exergy destruction with compression increases with the increase in compression ratio for all selected fuels. It is due to the reason that at high compression ratio, temperature of the gases increases which results in more heat transfer through the walls of the cylinder (assumed at 400K) and causes exergy destruction.

Exergy Lost with Internal Heat Transfer

The variations in exergy with internal heat transfer with the change in compression ratio are shown in figure 3 for all selected fuels. It is shown that exergy with internal heat transfer decreases with the increase in compression ratio for all selected fuels and it is because of the reason that combustion temperature increases with the increase in compression ratio which results in lower exergy destruction during heat transfer from the products (burned gas) to the reactants (unburned gas) inside the combustion chamber. Exergy with internal heat transfer with oxygenated fuels are slightly lower than the hydrocarbon fuels.

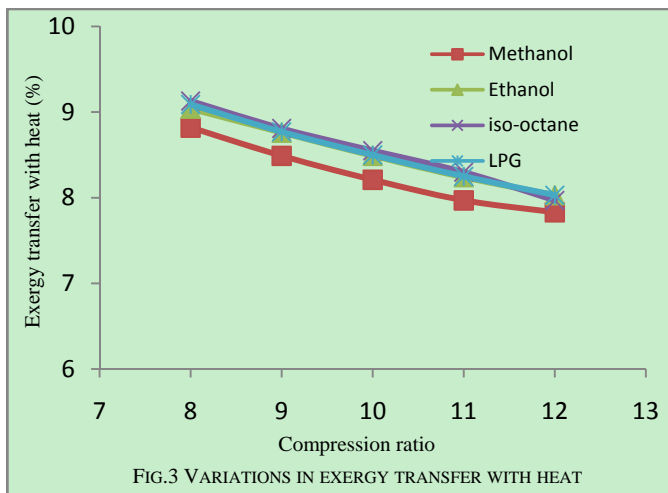


FIG.3 VARIATIONS IN EXERGY TRANSFER WITH HEAT

Exergy Destruction with Combustion

Fig. 4 shows the variations of exergy destruction during combustion with the change in compression ratio for all selected fuels. It is found that exergy destruction during combustion decreases with the increase in compression ratio for all selected fuels and this is due to the reason that reactants temperature increases with the increase in compression ratio so the combustion temperature increases which results in lower exergy destruction during combustion. The irreversibilities during combustion are highest for the iso-octane and it is because of the reason that iso-octane have long chain molecules and complex structure than other examined fuels.

It is also observed that among all examined fuels, the exergy destruction of methanol is minimum during combustion which is due to rich oxygen contents. Similar to this, the exergy destruction during

combustion for ethanol is also less than hydrocarbon fuels.

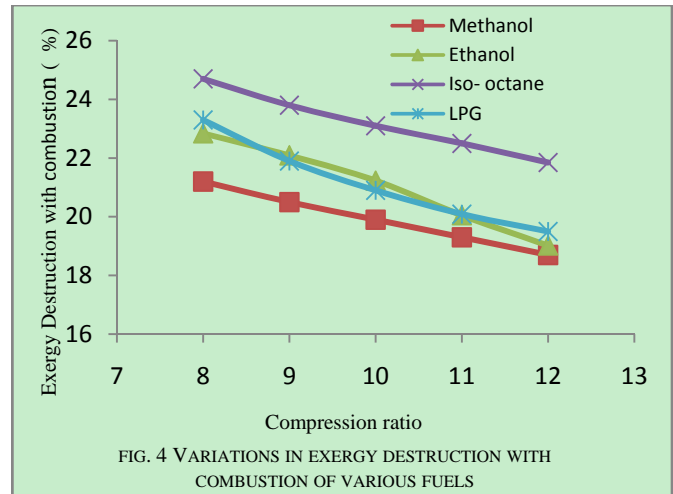


FIG. 4 VARIATIONS IN EXERGY DESTRUCTION WITH COMBUSTION OF VARIOUS FUELS

Exergy Destruction during Expansion Process

Fig. 5 shows the variations in exergy destruction during expansion with the change in compression ratio for all selected fuels. It is found that exergy destruction during expansion increases with the increase in compression ratio for all selected fuels. It is because of higher pressure and temperature after combustion which results in higher heat transfer through the walls during expansion of all fuels. It is also observed that Exergy destruction during expansion of oxygenated fuels are higher than the hydrocarbon fuels. This is due to the reason that oxygenated fuels have lower value of T_3 and T_4 than the hydrocarbon fuels which results in higher entropy generation.

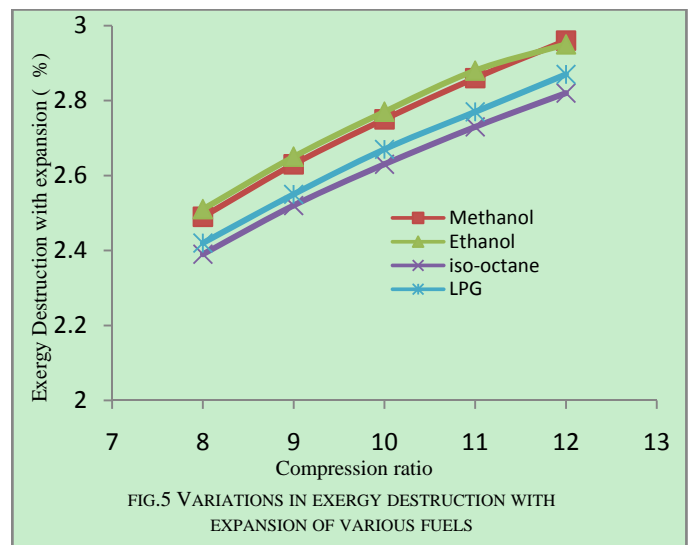
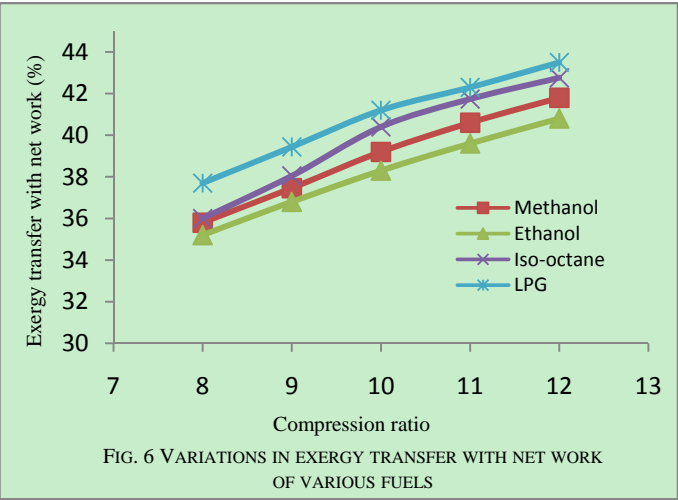


FIG.5 VARIATIONS IN EXERGY DESTRUCTION WITH EXPANSION OF VARIOUS FUELS

Exergy with Net Work

Fig. 6 shows the exergy with net work with the change in compression ratio for all selected fuels. It is shown that exergy transfer with net work increases with the increase in compression ratio for all examined fuels. This is due to reason that high compression increases the temperature and pressure after the combustion which results in increase in exergy transfer with net work. The exergy with net work are significantly lower in case of oxygenated fuels than the hydrocarbon fuels and it is because of the reason that higher pressure and temperature after combustion and lower adiabatic index during expansion.



It is also observed that exergy transfer with net work is lower in case of ethanol and highest for LPG among all examined fuels.

First Law Efficiency

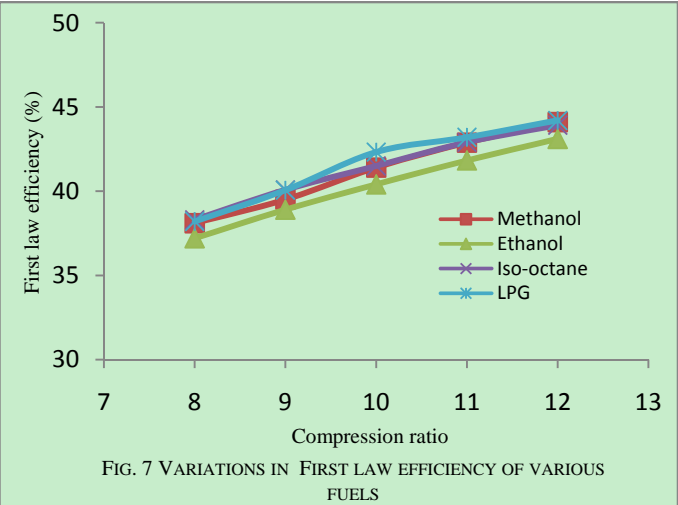
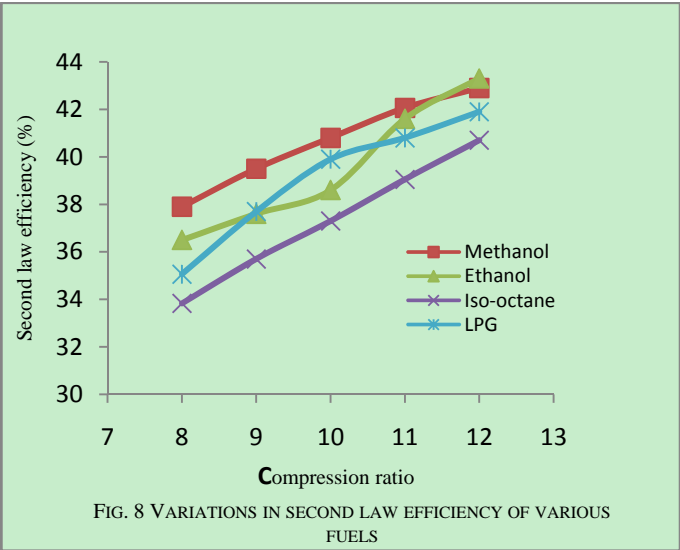


Fig. 7 shows the variations of first law efficiency with the change in compression ratio for all examined fuels. It is observed that first law efficiency increases with the increase in compression ratio for all examined fuels. This is due to the reason that with high compression ratio, pressure and temperature of the gases increases during combustion at constant volume. It is shown that first law efficiency of hydrocarbon fuels is slightly higher than the oxygenated fuels and this is due to reason that hydrocarbon fuels have higher pressure and temperature after combustion and high value of lower heating value. First law efficiency of methanol and iso-octane comes closer to each other between the compression ratios of 10 to 12. This is due to the reason that methanol have lower heating value and lowest combustion irreversibilities than the iso-octane. It is also shown that first law efficiency of ethanol is significantly lower among all selected fuels and it is because of lower heating value of ethanol.

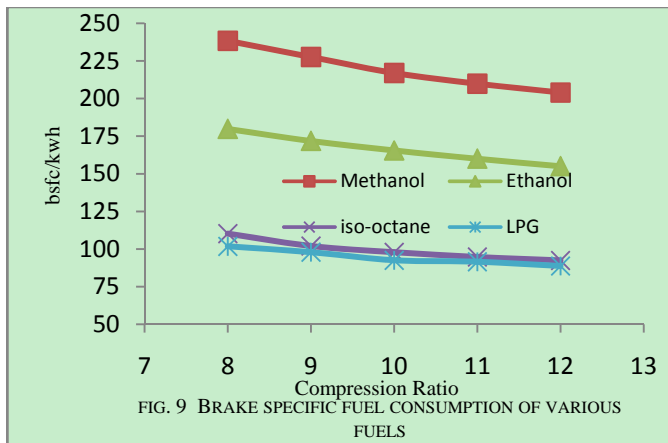
Second Law Efficiency



The variations in second law efficiency with the change in compression ratio are shown in figure 8 for all selected fuels. It is seen that second law efficiency increases with the increase in compression ratio for all selected fuels and this is due to the reason that combustion irreversibilities decreases with the increase in compression ratio. It is also observed that second law efficiency of oxygenated fuels is significantly higher than the hydrocarbon fuels. This is due to the reason that combustion irreversibilities and loss of exergy with exhaust gases are lower in oxygenated fuels.

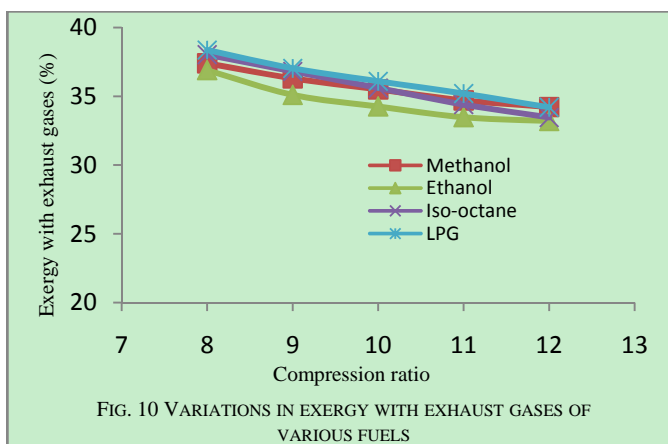
Brake Specific Fuel Consumption

The variations in brake specific fuel consumption per kwh (assumed 3000rpm) with the change in compression ratio is shown in figure 9 for all selected fuels. It is shown that brake specific fuel consumption decreases with the increase in compression ratio for all selected fuels. The brake specific fuel consumption is highest for methanol among all selected fuels.



This is because of reason that methanol has lower value of lower heating value among all selected fuels that means less amount of heat is generated during combustion of methanol. Similar to this, brake specific fuel consumption of ethanol is also higher than the hydrocarbon fuels because of lower heating value. It is also observed that the brake specific fuel consumption of iso-octane is lowest among all examined fuels and this is because of the reason that iso-octane has highest lower heating value among all selected fuels. Thus it is concluded that oxygenated fuels have higher specific fuel consumption than the hydrocarbon fuels.

Exergy with Exhaust Gases



The variations in exergy lost with exhaust gases with the change in compression ratio are shown in figure 10. It is shown that exergy lost with exhaust gases decreases significantly with the increase in compression ratio for all selected fuels. This is because of the reason that with high compression ratio, more amount of exergy is being carried with work because of higher temperature and pressure which results in lower temperature and pressure after expansion process. The exergy lost through exhaust gases of oxygenated fuels is less than the hydrocarbon fuels. This is because of the reason that the oxygenated fuels have lower temperature and pressure after expansion than the hydrocarbon fuels.

Ecological Efficiency

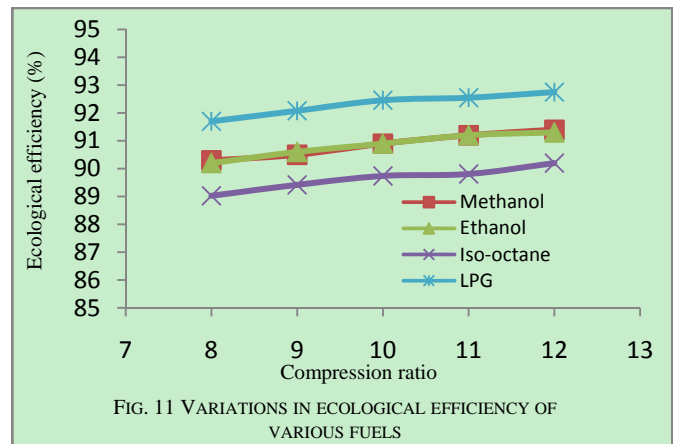
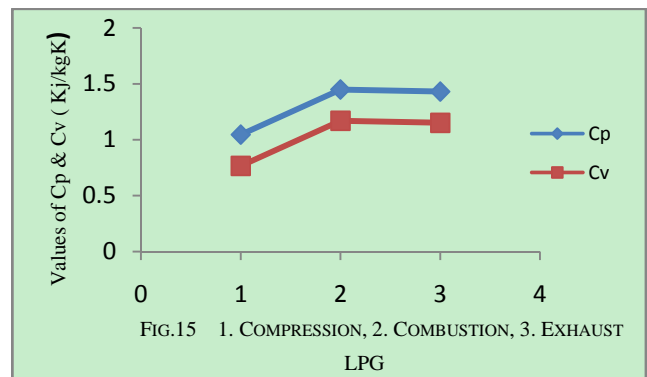
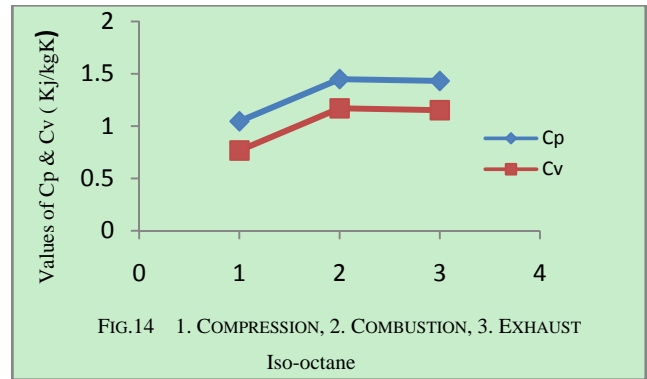
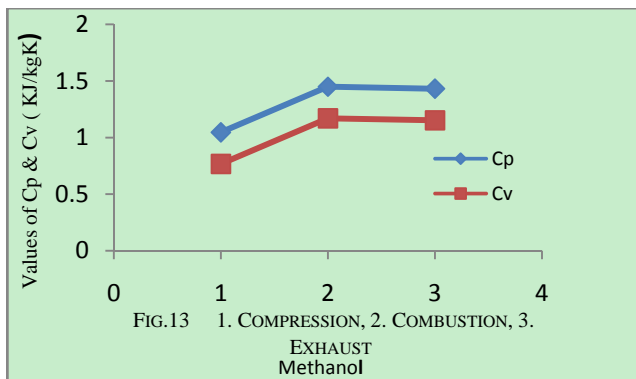
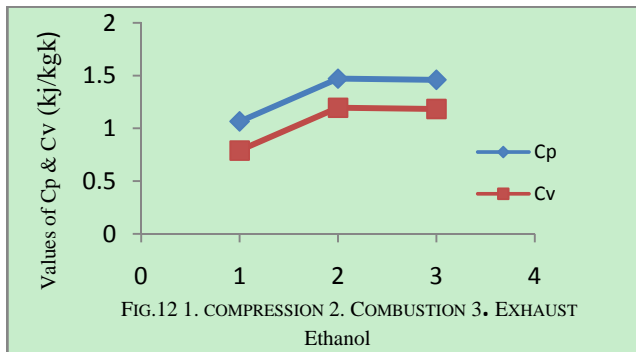


Fig. 11 shows the effect of the change in compression ratio on the ecological efficiency of the four examined fuels operated on SI cycle. It is found that ecological efficiency of all the fuels increases with the increase in compression ratio and it is because of the reason that with the increase in compression ratio, thermal efficiency increases. From the figure, it is also shown that ecological efficiency of LPG is significantly higher than among all the four fuel examined. It is due to the low value of pollutant indicator and higher value of motor octane number than the all selected fuels. Higher octane value of LPG produces better vehicle performance and less CO₂ emission. In SI engine cycle, LPG is used as a dry gas without lead as burning agent which results in particulate matter in exhaust. It is also observed that the ecological efficiency of the iso-octane is lowest among all the examined fuels and this is due to the reason that CO₂ emissions and particulate matter are significantly higher in the exhaust of iso-octane. The ecological efficiency of the oxygenated fuels that is ethanol and methanol is

higher than the iso-octane but lower than the LPG but very very close to each other and this is due to the reason that some amount of oxygen is present in the oxygenated fuels so these fuels burns smoothly and causes less CO₂ emissions.

Specific Heats of Different Fuels

Figure12, 13, 14, and 15 shows the change in specific heats at constant pressure and constant volume for the ethanol, methanol, iso-octane and LPG during compression, combustion and exhaust processes for the air fuel mixture. The values of specific heats increases with the increase in temperature of the air fuel mixture of all the fuels examined which results in more specific heats during combustion which is calculated at mean temperature of $(T_2+T_3) / 2$. The values of specific heats for the iso-octane and LPG are significantly lower than the ethanol and methanol during compression, combustion and exhaust. The specific heat of ethanol during compression is 0.59% higher than methanol and 1.9% greater than the iso-octane and LPG. The specific heats of methanol during combustion are 0.6%, 4.59% and 2.16% higher than ethanol, iso-octane and LPG respectively. The specific heats of methanol during exhaust are 2.2%, 5.4% and 4.1% higher than ethanol, iso-octane and LPG respectively.



Conclusions

In this study, the use of alternative fuels that is methanol, ethanol, iso-octane, and LPG in SI engine has been evaluated by using a thermodynamic model of exergy analysis.

The result shows that the first law efficiency second law efficiency and ecological efficiency increases with the increase in compression ratio for the all investigated fuels. Exergy destruction during compression and expansion increases with the increase in compression ratio. The results also show that the exergy destruction during combustion and exhaust decreases with the increase in compression ratio for all the examined fuels. Exergy with internal heat transfer decreases with the increase in compression ratio. The concept of ecological efficiency depends upon the environmental impact caused by CO₂, SO₂, NO_x, and particulate matter. The ecological efficiency of the LPG is 2.75% higher than the iso-octane and approximately 1.52% higher than the oxygenated fuels for the selected range of compression ratio. The second law efficiency of methanol is 3.69%, 10.7%, and 7.49% higher than the ethanol, iso-octane and LPG at a compression of 8 and this reduces to 1.0%, 5.12% and 2.33% at a compression of 12. Exergy with exhaust

gases is lowered by 1.63%, 5.07%, and 1.3% for the methanol, ethanol, and iso-octane respectively than the LPG. Thus the exhaust gases for LPG in case of spark ignition internal combustion engine are less polluting than the methanol, ethanol, and iso-octane. Exergy with net work is highest for the LPG and it increases with the increase in compression ratio for all the selected fuels. It is 5.76%, 7.93% and 2.88% greater than the methanol ethanol and iso-octane respectively. Exergy lost with internal heat transfer with iso-octane is slightly higher than the ethanol (0.7%), LPG (.58%) and sufficiently higher than the methanol (3.9%). Brake specific fuel consumption for iso-octane, ethanol, and methanol are higher, about 6.22%, 79.72% and 135.46% respectively, in comparison to iso-octane. The specific heats of all the fuels increase from compression to the combustion and decreases slightly during exhaust stroke. This variation in specific heats is taken into consideration while calculating energy distribution in different processes of otto cycle.

Nomenclature

E	exergy (kJ) of the fuel supplied
e_D	exergy destruction (kJ)
e_Q	exergy with heat transfer (kJ)
e	specific exergy (kJ/kg)
e_{physical}	physical exergy (kJ/kg)
e_{chemical}	chemical exergy(kJ/kg)
e_D	exergy destruction(kJ/kg)
ΔG	change in Gibb's function for reaction (kJ/kg)
LHV	lower calorific value (kJ/kg)
h	specific enthalpy (kJ/kg)
h_f	enthalpy of formation on molar basis (kJ/kg)
Δh	change in specific enthalpy (J/kg)
M	molar mass (kg/mol)
N	number of moles
n	polytropic index
P	pressure (bar)
r	compression ratio
Q_{in}	heat supplied during heat addition process (kJ/kg)
Q_{out}	heat rejected during exhaust process (kJ/kg)
\bar{R}	molar gas constant (kJ/K mol)
S	entropy (kJ/ kg K)

s	specific entropy (kJ/kg K)
T	absolute temperature (K)
T_o	ambient temperature (K)
V	volume (m ³)
v	specific volume (m ³ /kg)
w	work (kJ/kg)

Subscripts

Dest	destruction
e	equivalent
g	gas
i	inlet
e	exit
r	control volume
1	inlet
2	exit
P	pressure
V	volume
Gen	generation
f	fuel
comb	combustion
ph	physical
Q	heat
W	work

Greek and other symbols

η_I	first law efficiency
η_{II}	second law efficiency
γ	ratio of specific heat capacities

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